Environmental and Safety Impacts of HFC Emission Reduction Options for Air Conditioning and Heat Pump Systems

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ABSTRACT

Global warming emissions reduction strategies for Air Conditioners and Heat Pumps include (1) minimization of refrigerant emissions throughout the product lifecycle; (2) choice of refrigerant such as HFC, hydrocarbon and so on; (3) choice of refrigeration cycle or technology; (4) minimization of indirect CO_2 emissions (maximization of energy efficiency). In this paper we discuss all four of these options in relation to replacements for HCFC-22 in Air Conditioning and Heat Pump Systems. HCFC-22 is the most widely used fluorocarbon refrigerant in the world and is due for phase out according to stratospheric ozone depletion regulations in Article 2 countries early in the next century.

INTRODUCTION

In response to the international agreement, the Montreal Protocol on Substances that Deplete the Ozone Layer, that was signed in September 1987 and later amended, the production and consumption of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) came under control and will eventually be phased-out. With CFCs already phased-out of production in Article 2 countries, current attention is now being paid to replacements of HCFCs. The amendments of the Protocol in Copenhagen and Vienna called for a consumption “cap” or freeze of HCFCs production in 1996. This consumption “cap” value calculated from 1989 consumption data is scheduled to be reduced by 35% in 2004, with further cutbacks in later years until a 100% phase-out of all HCFCs occurs in 2030 in Article 2 countries. HCFC-22 has been the primary refrigerant for air conditioning and refrigeration for many years and continues to be used today as a replacement for CFCs in many applications.

HCFC-22 replacement options for air conditioners and heat pumps can be grouped into three categories. The first category is fluorocarbons that are used in conventional vapor compression cycles. The most popular choices are HFCs such as R-410A (a near-azeotropic mixture of HFC-32 and HFC-125), R-407C (a zeotropic blend of HFC-32, HFC-125 and HFC-134a), and R-134a. The second category is alternative working fluids in conventional vapor compression systems that include hydrocarbons such as propane (R-290) and ammonia (R-717). Finally the last category is alternative cycles that include absorption systems, transcritical (CO_2 or R-744), and air cycle. A discussion of each category follows.

The first category of replacement refrigerants requires the least change for component and system manufacturers as well as installation and service contractors. Alternative refrigerants that have similar safety and handling characteristics as R-22 (non-flammable, low toxicity) are employed. This would allow the continued use of the infrastructure now in place to manufacture, distribute, install, and service existing air conditioning and heat pump systems and make for a quicker, easier, and less costly transition to ozone benign working fluids.

The three HFC options have very different performance characteristics. R-134a is a lower capacity and lower pressure refrigerant than R-22. It would require greater compressor displacement and larger heat exchangers to match the capacity and efficiency of an R-22 air conditioner. At this point in time, it is only used in larger capacity a/c systems that utilize screw or centrifugal compressors. Manufacturers of smaller high volume a/c systems do not view R-134a favorably due to the added expense of building larger systems to handle the same duty as a comparable R-22 system.

R-410A is a higher capacity and higher pressure refrigerant. Systems that utilize this refrigerant would have smaller compressor displacements. When systems are optimized for this refrigerant, heat exchangers could be reduced in size and still achieve comparable capacity and efficiency as R-22. This refrigerant is currently regarded as the leading long-term refrigerant for residential and small commercial applications in North America and Japan as well as other parts of the world.

R-407C has capacity and pressure close to R-22. It is a zeotropic refrigerant with a temperature glide of approximately 6^{o}C (10 ^{o}F) which can lead to greater concerns about maintaining composition control during the distribution, installation, and servicing of this product. Unlike the other two candidates, this refrigerant can be used in either existing systems (requires some changes, e.g. oil) or in new systems that were originally designed for R-22. The efficiency of these systems is somewhat lower with this refrigerant (~5% or so lower than R-22), especially in systems designed for R-22. This refrigerant has seen considerable use in Europe lately due to the accelerated R-22 phase-out and the lack of time to re-design systems for the higher pressure refrigerant, R-410A.

EXAMINATION OF R-410A

Due to the interest in this refrigerant, a discussion of the characteristics, performance, and environmental impact of systems utilizing this refrigerant is in order. In order to quantify the performance of air conditioning systems with R-410A, an understanding of the performance of typical compressors designed to operate with this refrigerant is needed. In Figure 1 the efficiency (COP) of the R-410A compressor is compared to that of the comparable R-22 compressor. The conditions are 7^{o}C evaporator temperature, 11^{o}C superheat, and 8^{o}C subcooling. The performance is plotted as a function of condensing temperature. Also included is the thermodynamic efficiency as determined by REFPROP 6.0[1] and the compressor isentropic efficiency (ratio of compressor efficiency to thermodynamic efficiency). The compressor efficiency was determined from an average of the performance of a series of R-410A scroll compressors from a major compressor manufacturer[2].

At moderate and low condensing temperatures, the compressor performance makes up for some of the decrease in thermodynamic efficiency of R-410A relative to R-22 (i.e.

the isentropic efficiency of the R-410A compressor is higher than the R-22 compressor at these conditions). At high condensing temperatures (> 47°C) this is not the case.

Performance based on compressor data alone assumes that the saturated suction temperature and saturated discharge temperature are identical for R-410A and R-22. With changes in heat exchanger performance due to differences in the transport properties and pressure, this is not the case in real systems. In order to quantify these differences, information on the heat transfer and pressure drop performance of both refrigerants is necessary.

Data from three studies[3,4,5] on evaporative heat transfer and pressure drop and four studies[3,4,5,6] on condensing heat transfer and pressure drop was summarized. All these studies utilized 9.5mm (3/8") O.D. tubes without internal enhancements and covered similar ranges of mass fluxes. The data was averaged and is presented in Figures 2 through 5.

The pressure drop per equivalent length is less for R-410A than R-22 (Figures 2 & 4) at a given mass flux. Using a representative evaporator mass flux for an R-22 system of 200 kg/sec-m^2, an increase to ~280 kg/sec-m^2 will result in the equivalent pressure drop for R-410A. Since the pressure is higher for R-410A, it will require a 50% increase in pressure drop to get the equivalent saturation temperature change as R-22. Therefore, the mass flux could increase to ~340 kg/sec-m^2 in an R-410A system and result in the same impact on saturation temperature as the original 200 kg/sec-m^2 did for R-22. Examining Figure 3 shows the impact of the higher mass fluxes on the evaporation heat transfer coefficient. At the same mass flux there is an increase of 55% in the refrigerant side heat transfer. At the equivalent pressure drop there is a 90% increase in heat transfer and at the equivalent saturation temperature drop there is a 115% increase in heat transfer. Although the condensing heat transfer (Figure 5) for R-410A is only slightly higher than R-22 at the same mass flux (~15%), at the mass flux associated with the equivalent pressure drop it is 35% higher and at the mass flux associated with the equivalent temperature drop it is 65% higher.

To determine the impact of the heat transfer and pressure drop characteristics on system performance, a representative 12 kW split-system air conditioner was modeled. Heat exchanger geometry and airside performance was taken from computer runs of the PUREZ heat pump model program[7]. The overall heat transfer coefficient was determined from the airside coefficient produced by the PUREZ model and the information on refrigerant side discussed in the previous section. The NTU and the effectiveness of both heat exchangers were than calculated assuming cross-flow heat exchange. Optimum mass flow rates that produced the highest evaporator outlet temperature and the lowest condenser inlet temperature were determined. The optimum evaporator flow rate was ~25% higher for R-410A and the condenser was optimum was 35% higher.

Overall system performance was then determined by reducing the evaporator outlet temperature and increasing the condenser inlet temperature by ~1^{o}C for R-22 to take into account the pressure drop in the inter-connecting lines (the line loss was 0.7 ^{o}C for R-410A due to the lower impact of pressure drop). This resulted in the determination of the saturated suction and saturated discharge temperatures of both the R-22 and R-410A system compressor. Using the appropriate 10 point performance curve fit, compressor performance was determined for each system. To get system capacity, the compressor capacity was reduced by 300 watts due to the heat added by the indoor air blower. To get system power consumption, the compressor power was increased by 600 watts due to the blower and fan power consumption. System COP was determined by dividing the system capacity by the system power consumption. At typical design conditions (35^{o}C ambient) and at average operating conditions (28^{o}C ambient), the efficiency of the R-410A system is ~2 and 5% higher in efficiency than the comparable R-22 system. This is consistent with results of actual system tests of soft-optimized systems[8]. Again this was due to the higher saturated evaporator temperature and the lower condenser saturated temperature of R-410A relative to R-22.

When designing new systems with R-410A, the improved heat transfer and pressure drop properties will impact heat exchanger design by reducing the surface area required to meet the targeted efficiency. This will reduce the cost of the heat exchanger needed to meet this efficiency and reduce the internal volume of the system thereby reducing refrigerant charge. In addition, due to the reduced sensitivity to pressure drop, optimum tube diameters will be smaller for R-410A than R-22, a further reduction in refrigerant charge. Another property that reduces the required amount of refrigerant is the lower liquid density of the refrigerant. This leads to a 12% reduction in the weight of the refrigerant required. The total of these effects is an overall reduction of 25 to 30% charge reduction in fully optimized R-410A systems.

ALTERNATIVE FLUIDS AND CYCLES

The second and third categories of replacements require additional developments in order for a safe and economical transition to be made. In the case of using a flammable and/or more toxic working fluid such as propane or ammonia safeguards must be put in place to avoid leakage of the refrigerant into the occupied space. Although ammonia, long used in industrial refrigeration systems is generally not considered for residential a/c systems. In addition to the higher toxicity characteristics, the incompatibility with copper and common compressor designs (i.e. hermetic compressor/motor systems) limits its application. Hydrocarbons such as propane may also be used in conventional vapor compression systems. However the flammability of these materials will also limit their applications. In anything other than very small systems (e.g. refrigerators) the amount of refrigerant will warrant the elimination of any part of the refrigerant containing system from the occupied space. A possible solution is to use a secondary heat transfer loop that circulates a non-flammable fluid from an outdoor unit that contains the flammable refrigerant to the indoor air distribution system. However this significantly adds to the cost and also reduces the efficiency due to the added heat exchanger temperature difference needed to run the secondary loop.

In addition to conventional vapor compression systems, other technologies have been suggested for air conditioners and heat pumps. Absorption systems have long been used for this purpose in large central chiller systems. These generally suffer from higher first cost and lower operating efficiencies and are usually only justified when there is a significantly higher cost of electricity as compared to the cost of natural gas. Development of smaller, efficient , cost-effective equipment is not a near-term likelihood.

There is presently a good deal of interest in transcritical CO_2 systems. However at the present time, the efficiency of these systems are considerably lower than conventional vapor compression systems and affordable compressor technology remains an open issue.

CONCLUSIONS

When examining the environmental impact of a/c and heat pump systems it has been shown repeatedly that the greatest contribution to global warming comes from the power it takes to run the system over its useful life. Oak Ridge National Laboratory published a comprehensive study in 1997[9] that compares the contribution of the direct impact of the refrigerant due to leakage to the impact due to the power the device consumes. The average contribution to the total global warming due to refrigerant leakage averages 3 to 4%. It also shows the benefit of increasing the efficiency of the system. A 20% increase in efficiency translates to a 12% decrease in total warming or more than three times the total direct effect of the refrigerant.

In order to reduce the global warming influence of future air conditioning and heat pump systems increasing the energy efficiency of these systems will provide the greatest impact. It was shown that using R-410A would allow for a more cost-effective approach when designing high efficiency air conditioners and heat pumps. It would reduce the size increase of heat exchangers needed for high efficiency equipment. Optimized design will also reduce the required refrigerant charge by 25 to 30% thereby reducing the direct effect as well.

Other leading choices such as hydrocarbons and CO_2 make designing higher efficiency equipment more difficult. In the case of hydrocarbons, the need for a secondary loop or other changes in the system necessary to reduce the risk of a highly flammable refrigerant use materials and thereby cost that could be used to improve the efficiency of non-flammable refrigerant systems. Improvements are also needed in CO_2 systems to make-up for the considerably lower thermodynamic efficiency of the cycle.

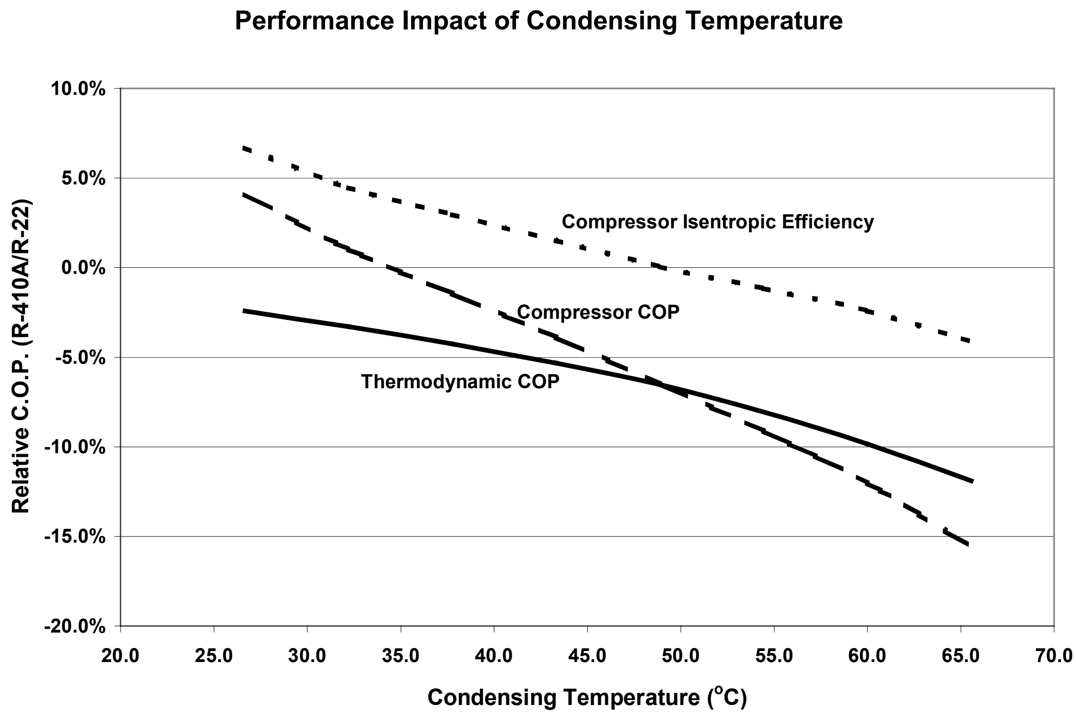

Figure 1

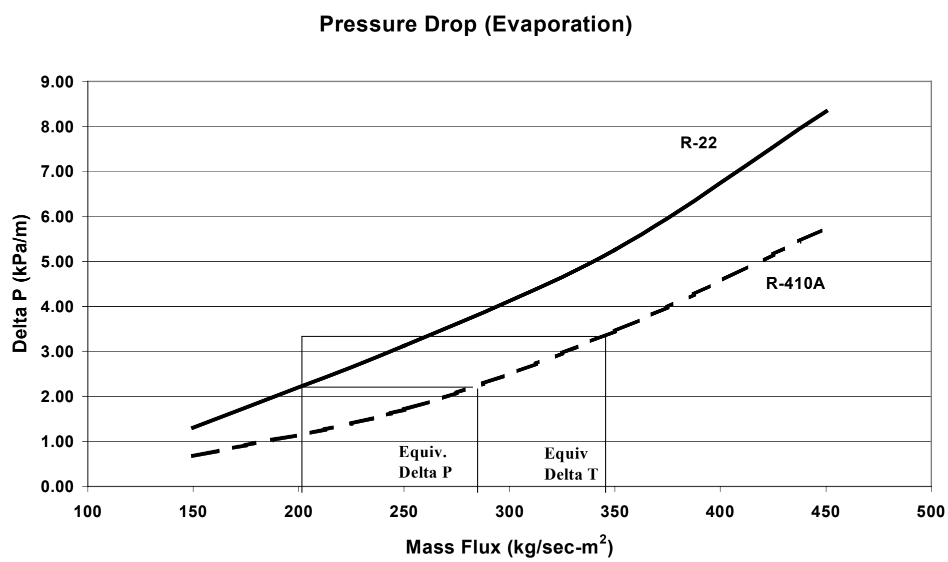

Figure 2

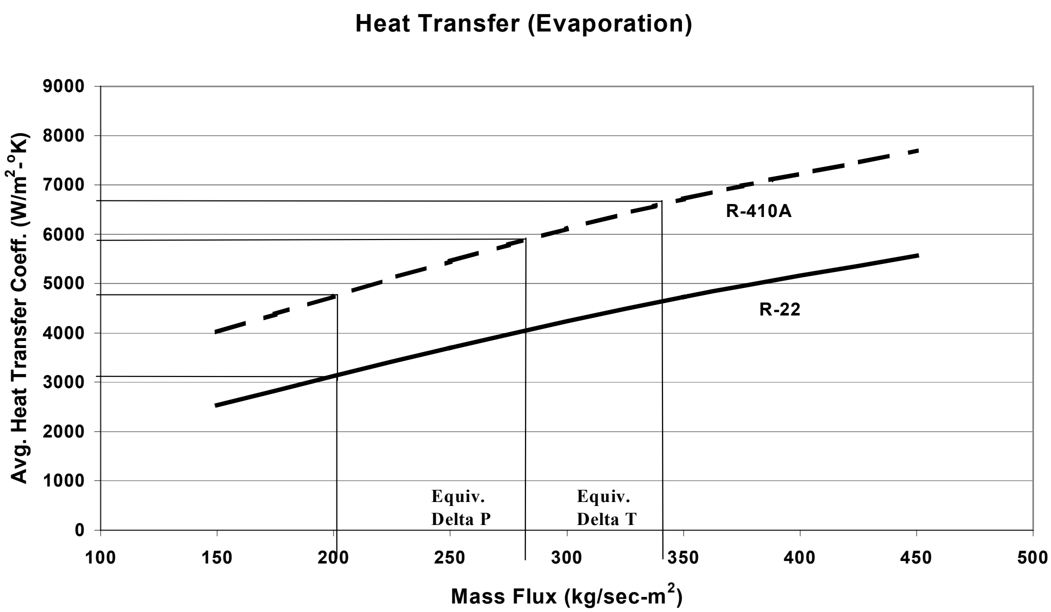

Figure 3

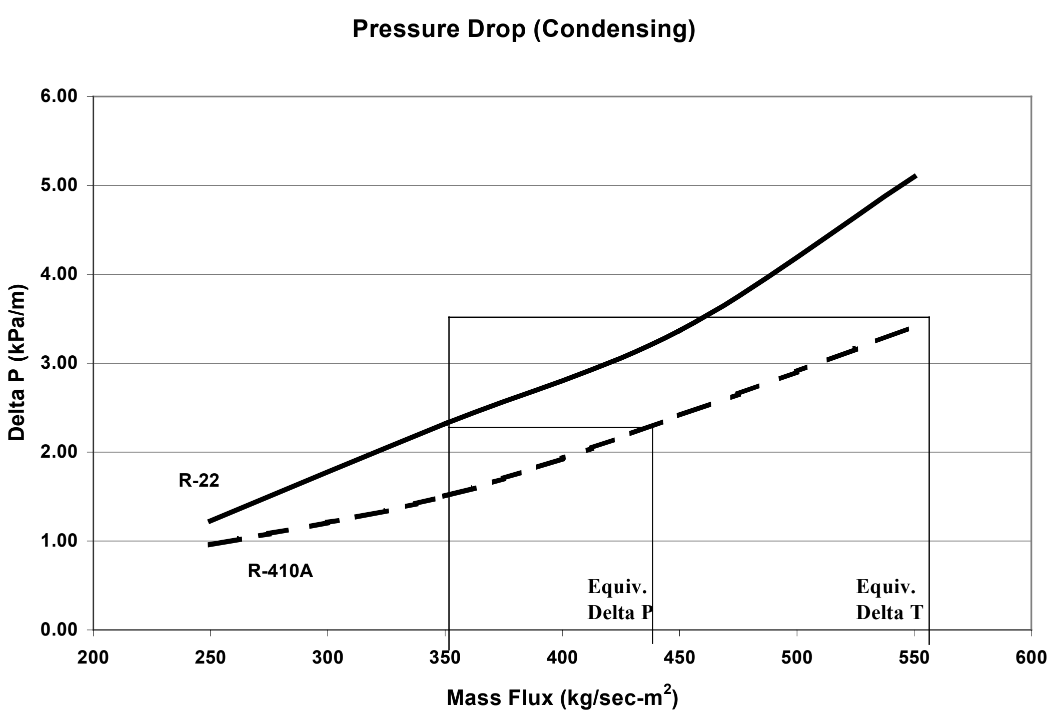

Figure 4

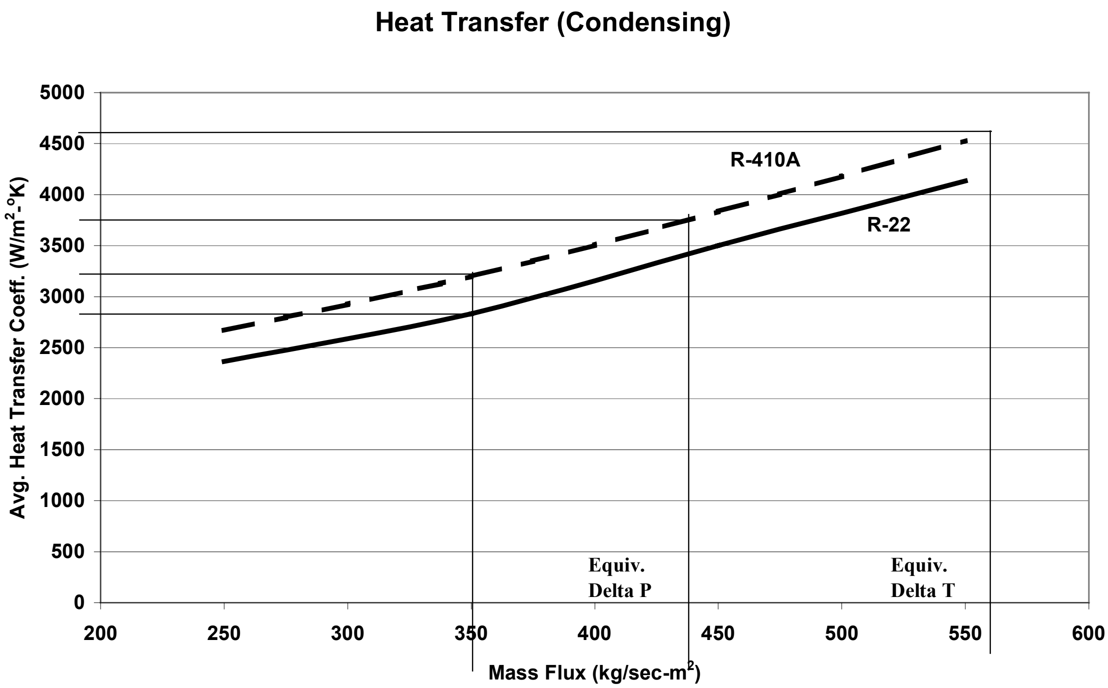
Heat Transfer (Condensing)
5000
4500
4000
3500
3000
2500
2000
1500
1000
500
0
Avg. Heat Transfer Coeff. (W/m2-oK)
R-410A
R-22
Equiv.
Delta P
Equiv.
Delta T
200
250
300
350
400
450
500
550
600
Mass Flux (kg/sec-m2)

Figure 5